

An Investigation into the Accuracy of Orifice Based Flow Estimates for Pulsating Compressible Flows

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ABSTRACT

Orifices, flow nozzles and arbitrarily shaped flow obstructing flow measurement devices are widely used to estimate EGR flow rates in engines, and also used to model flow restricting components like valves in engine analysis tools such as GT-Power. The standard assumptions about the flow discharge coefficient and its variation with Reynolds number are based on investigations of orifices across steady non-pulsating flows, widely reported in literature. In this work, the discharge coefficient for steady state pulsating flow as well as accelerating pulsating flow, commonly encountered during steady state and dynamic engine operation respectively, were investigated by installing an orifice on the exhaust side of a naturally aspirated diesel engine, while making reference flow measurements with a Laminar Flow Element on the intake side. 'Snap Throttle' tests were performed to accelerate the flow on the exhaust side with a sudden increase in exhaust gas temperature and accompanying decrease in density. Contrary to reported literature based on steady non-pulsating flow, a linear and directly proportional relationship between coefficient of discharge and Reynolds number was determined for steady state operation at Reynolds numbers exceeding 45,000. The coefficient of discharge changed by about 11% as the Reynolds number increased from approximately 45,000 to 103,000, corresponding to part load and full load steady state conditions at three engine speeds. Compressibility effects were observed to be important. During dynamic operation, the primary source of inaccuracy for the orifice based estimate was determined to be the slow temperature measurement rather than dynamic pressure waves corresponding to the acceleration of flow during the snap throttle event. Another objective of this work was to understand the role of factors other than the fuel-Oxygen ratio in the production of smoke spikes during dynamic operation. Such factors include injector dribbling/overflow caused by pulsations in the fuel injection system, wall impingement, lack of mixture preparation while the full load in-cylinder flow field is established, and the relatively low cylinder wall temperature during the initial period of full load operation. Engine opacity during the snap throttle event was observed to track the fuel-Oxygen ratio, suggesting that these factors were of secondary importance.

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INTRODUCTION AND BACKGROUND

Previous work by the author attempting to understand the fundamental reason causing smoke spikes during dynamic operation in electronically controlled turbocharged diesel engines with adjustable fuel-Oxygen limits, has shown that the in-cylinder fuel-Oxygen ratio estimate could be significantly inaccurate during dynamic operation [1-2]. [Figure 1](#) shows a transient segment containing one smoke spike investigated in that work. A GT-Power model calibrated with steady state flow rate data was used to predict dynamic flow rates for the transient segment. These predictions were compared to the ECM estimates. Three underlying reasons for discrepancies between ECM estimates and modeled fuel-Oxygen ratios were reported: volumetric efficiency overestimation, cylinder-to-cylinder variation and EGR flow underestimation, by the ECM during dynamic operation.

This last reason was the predominant factor and resulted from inaccurate orifice ΔP based EGR flow estimates during the turbocharger lag period. The coefficient of discharge (C_d) of orifices or flow nozzles is generally determined at steady state operation. There is very little literature investigating the validity C_d for dynamic flow, e.g. work by Daily et al. done in the 50s [3]. For steady flow, even though the relationship between C_d and Reynolds number is well known, the author could not find any work investigating the validity of this relationship for pulsating compressible flow, or flows at high temperatures typical of flow through the EGR system. The current work is an attempt to understand orifice based measurement of pulsating engine exhaust gases in particular, during steady state and dynamic operation.

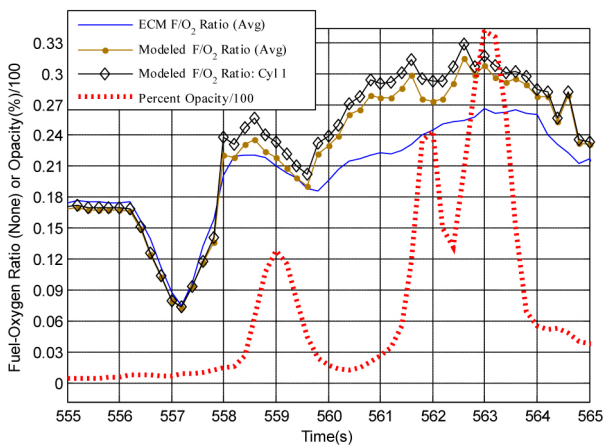


Figure 1. This opacity spike occurring during the turbocharger lag period was previously investigated by the author and found to be a result of high fuel-Oxygen ratio [1]. ECM estimated fuel-Oxygen was lower because of inaccurate orifice based EGR flow estimates. Fuel-Oxygen 'limiters' were therefore ineffective at preventing smoke spikes.

Because of the ubiquity of orifice plates as a cheap method of flow measurement, the steady state coefficient of discharge for orifice plates as a function of orifice geometry, pressure tap location and Reynolds number has been extensively studied since the 1930's [4] through the 70's [5,6]. The results have been reported in standard textbooks of fluid dynamics, e.g. Crowe et al. [7] and textbooks of engineering experimentation, e.g. Holman [8], as well as widely used fluid measurement handbooks, e.g. by Miller [9], Cusick [10] and Bean [11]. Investigations of Orifice flow in special situations such as highly viscous low Reynolds number flows seen in pipelines, e.g. Hollingshead et al. [12] have continued to the present day. Although the bulk of orifice investigations have been performed with incompressible liquids, orifice based methods for compressible flows are well established and standardized [13-14]. Many of the above mentioned works characterize the orifice C_d as a function of Reynolds number, but no such relationship for *pulsating compressible flow*, typical of engines, is available. The current work is an attempt to characterize orifice C_d as a function of Reynolds number for steady pulsating compressible flow, and to examine the validity of such C_d values for accelerating flows, encountered during dynamic operation of the engine.

It is common practice to use indirect CO_2 measurements to measure EGR flow in a test cell, and use it to calibrate orifice C_d . Subsequent orifice estimates are impossible to confirm directly during dynamic operations, unless highly specialized fast CO_2 measurements are made [15]. A direct measurement of the dynamic flow rate of hot EGR gases with high particle concentration is difficult. However if the orifice is placed in the exhaust stream, a direct flow rate measurement can be made of the identical fresh air flow going into the intake system. A square edged orifice plate was therefore installed downstream of the exhaust manifold as shown in figure 2:

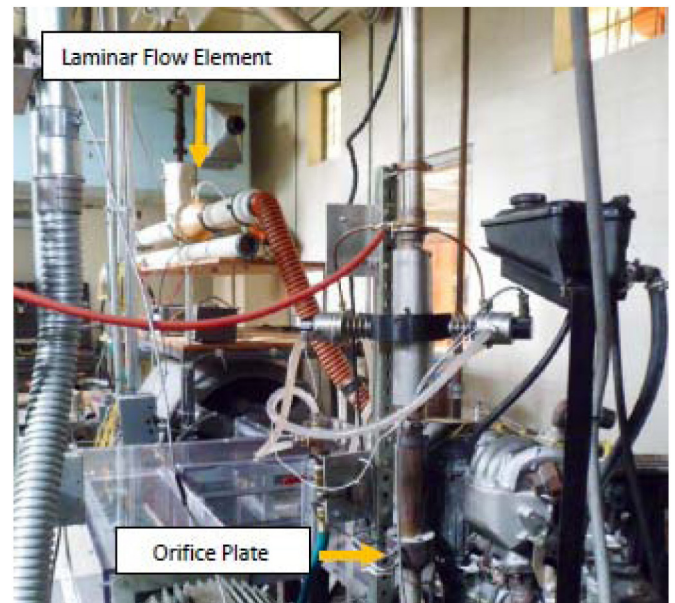


Figure 2. The orifice plate was installed downstream of the exhaust manifold, so that a direct measurement of flow rate could be made by an LFE located upstream of the intake manifold.

The flow entering the engine, measured accurately by a laminar flow element (LFE) plus the fuel injection rate was equal to the exhaust flow exiting the engine. A naturally aspirated 4 cylinder 'TUDS 5' Peugeot diesel engine with a mechanical fuel injection system was used for this work.

To test orifice based estimates for accelerating flow, 'snap throttle' tests were performed, wherein the throttle position was increased from 50% to 100% in about 2.5 seconds; the minimum time period allowed by the dynamometer controller. Exhaust gas velocity changed by 30%-40% at the orifice because of the increased temperature and reduced density of the exhaust gases at full load. The velocity on the intake side changed very gradually, decreasing slowly as engine cylinder wall temperatures increased and volumetric efficiency decreased at full load. The mass flow rate through the engine mirrored the intake velocity. The gradual change in intake velocity allowed the LFE mass flow rates measurement to be used as a reference. Figure 3 compares the drastic change in exhaust velocity at the orifice with the gradual change in intake velocity at LFE entrance during the snap throttle event.

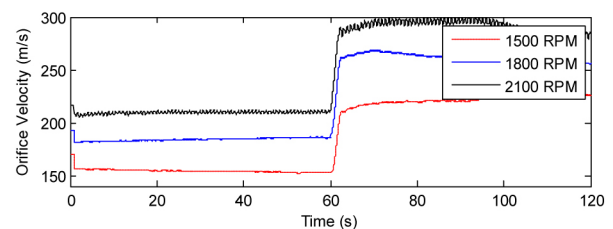


Figure 3. The gradual change in velocity at the LFE entrance (bottom subplot) allowed the LFE mass flow measurements to be used as a reference for the Orifice mass flow estimates, made while the flow accelerated rapidly at the Orifice (top subplot). Note that the exhaust density decreased in the opposite direction, such that the mass flow rate mirrored the LFE velocity.

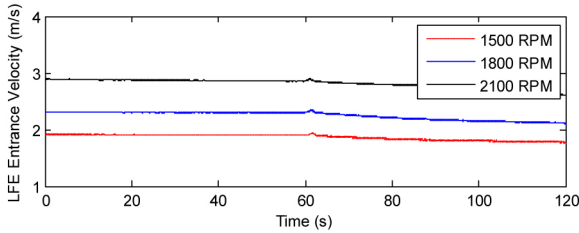


Figure 3. (cont.) The gradual change in velocity at the LFE entrance (bottom subplot) allowed the LFE mass flow measurements to be used as a reference for the Orifice mass flow estimates, made while the flow accelerated rapidly at the Orifice (top subplot). Note that the exhaust density decreased in the opposite direction, such that the mass flow rate mirrored the LFE velocity.

In addition to producing verifiable orifice based flow rate estimates, installing the orifice on the exhaust side also resulted in a reduction in the fresh air flow through the engine during steady state operation. The diameter of the square edged orifice (0.5 Inch, $\beta=0.31$) was chosen to increase the fuel-Oxygen ratio at 100% throttle to about 0.25 (see fig 18, 19, 20). Turbocharged engines operate at similar fuel-Oxygen ratios during the turbocharger lag period (see fig 1). The secondary purpose of this work was to understand the role of high fuel-Oxygen ratio relative to other factors (such as mixture preparation, injector dribbling, wall impingement) in the production of dynamic smoke spikes. Unlike a turbocharged engine where the fuel-oxygen ratio is quickly restored as boost builds up, the naturally aspirated engine in the current work retained the high fuel-Oxygen ratio after the throttle was snapped. If the exhaust opacity (smoke) during the rapid change in throttle position was significantly different than the opacity measured after the transition, then the role of the other factors could be considered significant. However this turned out not to be the case and the opacity simply tracked the fuel-Oxygen ratio. The exhaust opacity was measured with a 'Celesco' opacity meter, which employs the light extinction method. A laser beam is sent across the exhaust stream, and the percent loss of light intensity (opacity) is considered to be indicative of the 'smokiness' of exhaust.

RESULTS AND DISCUSSION

C_d at Steady State

Figure 4 shows the variation of C_d with Reynolds number calculated for the three engine speeds investigated: 1500 RPM, 1800 RPM and 2100 RPM. The coefficient of discharge was calculated with the well-known orifice discharge equation [8]:

$$\dot{m} = YMC_d A \sqrt{2\rho\Delta P_{\text{orifice}}} \quad (1)$$

where $\dot{m}$ is the 'true' mass flow rate measured by the Laminar Flow Element, M is the constant velocity approach factor given by equation (2), Y is the compressibility factor given by

equation (3), A is the area of the orifice, ρ is the density of the gas upstream of the orifice and $\Delta P_{\text{orifice}}$ is the pressure different across the Orifice.

The velocity approach factor M is a consequence of grouping of the area terms when deriving equation (1) from Bernoulli's principle. It depends solely on the ratio of orifice to pipe diameter β , and is constant for any given geometry:

$$M = \frac{1}{\sqrt{1 - \beta^4}} \quad (2)$$

The compressibility factor Y accounts for compressibility effects (Bernoulli's equation assumes incompressible flow). It is given by:

$$Y = 1 - \left(\frac{0.41 + 0.35\beta^4}{k} \right) \frac{\Delta P_{\text{orifice}}}{P_1} \quad (3)$$

where k is the specific heat ratio and P_1 is the pressure upstream of the orifice. Equation (3) is empirical, and is commonly used for orifice plates with flange taps [8].

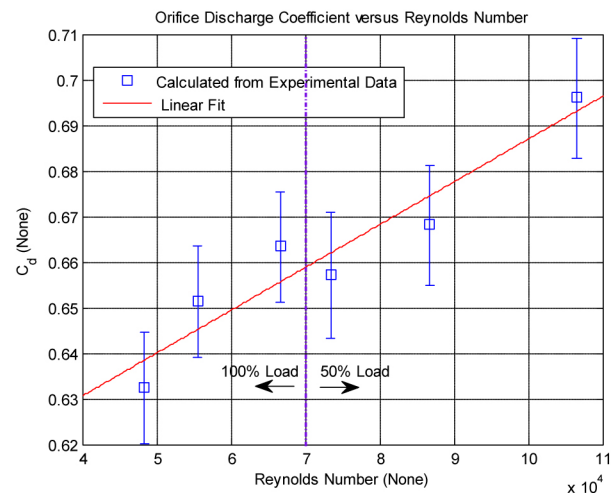


Figure 4. A continuous linear relationship was observed between C_d and Reynolds number, beyond $Re = 45,000$, contrary to widely used standard results for non-pulsating, compressible flow results not obtained from an engine, that claim a nearly constant C_d beyond a similar Re , for $\beta=0.31$. Here Re is based on the orifice diameter. Re based on pipe diameter, i.e. Re_D is lower by a factor of $\beta=0.31$. The lowest Re_D in this figure is ~14000, but the ASME handbook [13] reports constant C_d at Re_D exceeding 10,000.

It can be seen from figure 4 that C_d increases continuously with Reynolds number in a linear manner across a wide range of Reynolds numbers. It increases by about 13% as the Reynolds number increases from about 45,000 to 103,000. The 100% load points have lower Reynolds numbers because of the increased viscosity at higher exhaust temperatures. The viscosity was modeled with the empirical Sutherland equation [16] for air.